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MECHANICAL AND ENVIRONMENTAL
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LONG TERM INTEGRITY FOR SPACE STATION POWER SYSTEMS

by

F. A. Leckie
Mechanical & Environmental Engineering Department
University of California, Santa Barbara

and

D. L. Marriott
Department of Mechanical Engineering
University of Illinois

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CONTENTS

EXECUTIVE SUMMARY

1. A study has been made of the High Temperature Design Codes ASME N47, British R5, and the French RCC-MR Rules.

2. It is concluded that all these Codes provide a good basis of design for space application. The new British R5 is the most complete since it deals with the problem of defects. The ASME N47 has been subjected longer to practical application and scrutiny.

3. A draft code is introduced, "A Proposed Draft for High Temperature Design," in which attempts have been made to identify gaps and improvements are suggested.

4. The design appears to be is limited by creep characteristics. In these circumstances life is strongly affected by the selected value of the Factor of Safety. The factor of safety of primary loads adopted in the Codes is 1.5 Maybe a lower value of 1.25 is permissible for use in space.

5. Long term creep rupture data for HAYNES 188 is deficient and it is suggested that extrapolation methods be investigated.

LONG TERM INTEGRITY FOR SPACE STATION POWER SYSTEMS

1. INTRODUCTION

The operation of the N.A.S.A. Freedom Spacelab is dependent on the continuous availability of 25 kw of power. One element of the station is the novel form of solar power plant which uses the melting and freezing of a eutectic mixture of LiF-CaFe salts as a thermal storage medium (Fig. 1).

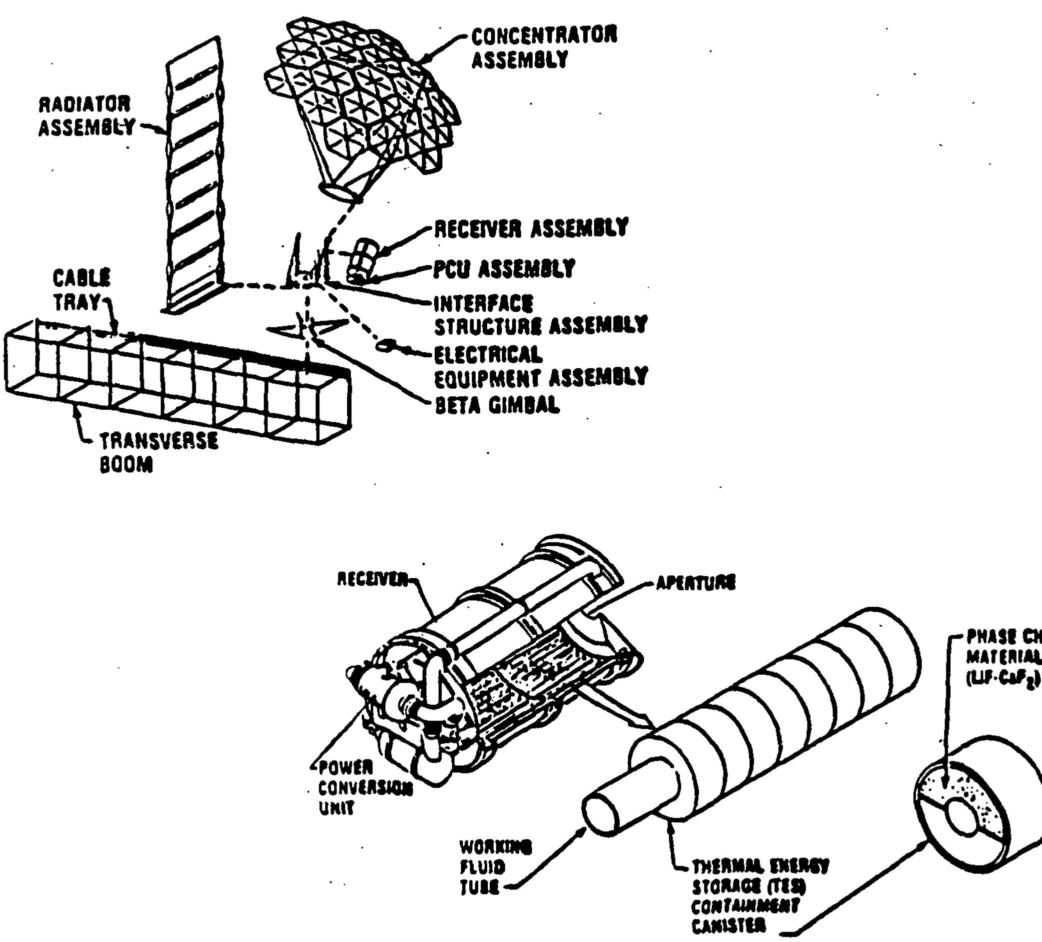

Figure 1. Space Station Freedom Solar Dynamic Power Module

The temperatures are very high in the region of 750^{o}C and the uninspected service life of the system is 30 years. The system combines the operating temperature of a gas turbine the life expectancy of fossil power plant, while it is inaccessible for inspection and repair, a circumstance which places the system far outside normal experience. Furthermore, the temperature range and the continuous cyclic conditions experienced by the Freedom Solar Storage Module define to a new application regime which might be termed "Very High Temperature Design" because the temperatures are substantially higher than normal in fossil and nuclear power plant while the life times without inspection are fifty times greater than those experienced by gas turbines.

The operating conditions of aircraft engines often defines the extremities of technical feasibility and the design procedure is supported by substantial statistical experience. This philosophy is appropriate when the production of a significant number of units is envisioned and experience with the new design adds to the statistical base. Furthermore, regular maintenance is an essential element of successful operation when the initiation and growth of flaws can be measured and decisions on replacement can be made on a timely and economic basis.

In fossil and nuclear power plant regular inspection is rigorously pursued as a precaution against severe accident. Normally the investment in such a plant is so large that each plant is unique and while the statistical knowledge of similar plants can be helpful in design, it does not ease the uncertainties associated with a "one-off" design. For these reasons the introduction of a Factor of Safety into the design of such plant is usual and indeed mandatory. It is normal to select the level of the working load to be 2/3 of the maximum load carrying capacity. The proof test for acceptance is usually 4/3 of the working load or 8/9 of the maximum load. These are the load levels adopted by a well-developed technology and their consequences are worthy of comment. It is common for power plant to have a design operating life of 15 years so that their design is limited by the creep deformation. For these conditions the failure life t_f can be expressed in the form

$$t_f = k\sigma^n$$

where k is a material dependent property and σ is a so-called reference stress which is proportional to the primary load applied to the component and n is a material constant, which may be about 5. Hence, if the design life is t_D the corresponding design stress σ_D is defined by the equation

$$t_D = k\sigma_D{}^n$$

The working stress σ_w assuming a factor of safety of 1.5 is then

$$\sigma_w = \frac{2}{3}\sigma_D$$

The corresponding estimated life of the component t_c is then

$$t_c = k\left(\frac{2}{3}\sigma_D\right)^n$$

so that the ratio of the estimated to design life is

$$\frac{\tau_c}{\tau_D} = \left(\frac{3}{2}\right)^n$$

If, for example, n=5 the ratio becomes

$$\frac{\tau_c}{\tau_D} = 7.6$$

so that the <u>Actual Life</u> is 7.6 times the <u>Design Life</u>.

It is of little surprise, therefore, that after operating for their Design Life these plants often show little signs of damage or distress. There is considerable interest currently to extend the life operating of apparently undamaged plant. Suppose for sake of argument that it is decided to extend the working life by a factor of two. The effect of this decision is to reduce the Factor of Safety from 1.5 to $\frac{1.5}{2^{1/5}} = 1.31$. It would appear that this reduction of the factor of safety is sufficiently small to justify extending working life of plant. Such decision are normally accompanied by plans to increase inspection frequency.

The uniqueness of the design of the Freedom Spaceship and the need for reliable performance over a 30 year life without maintenance places challenges on the designer which are outside normal experience. Previous discussion suggests that the experience gained from fossil and nuclear power plant could well provide a base of knowledge which could be directed with profit to the design of the power plant of the Freedom Space Lab.

The purpose of this study is to establish the current status of existing High Temperature Design Codes and to determine how the existing knowledge can be used to identify those gaps which must be closed before a reliable design procedure for Very High Temperature components can be established. It is hoped that the study is sufficiently general for the recommendations to be applicable not only to the solar power plant proposed for the space station "Freedom" but to any other types of space power plant .

2. STUDY OF EXISTING DESIGN CODES

Recommended procedures for high temperature design have remained sensibly static for more than 20 years, despite work during the intervening period on several time-dependent phenomena such as creep cracking, creep/fatigue interactions and creep ratchetting. In fact, for many years only one generally available guideline for high temperature design has existed outside the proprietary procedures used in specialist companies like gas turbine manufacturers. This guideline is ASME Code Case N47. Recently, however, there has been some forward movement with the publication of new design procedures, notably the British R5 route, and the French RCC-MR Rules.

N47 has been the virtual mainstay of high temperature design for many years, and provides many ideas which still form the framework for more recent developments. A review of this guideline is a good place to start.

2.1 ASME Code Case N47 (1)

Code Case N47 is the part of the ASME Boiler and Pressure Vessel Design Code relating to design for high temperature operation. It is a voluntary procedure. ASME Code practice is to try out a design procedure as a Code Case until consensus has been reached by experience in the field. Only then is it adopted as a mandatory procedure. After about 20 years of use, N47 has still not advanced to that point. However, the lack of any alternative has meant that N47 has a de facto status as a design procedure which is almost as strong as the mandatory sections of the ASME Code itself.

It is not proposed to review the Code Case completely. A summary of its contributions and shortcomings follow.

Stress Categorization

N47 follows the main ASME Code in partitioning total stresses into primary membranes and bending (Pm, Pb), secondary (Q) and Peak (F) components.

> *Primary* (Pm+Pb) stresses are load controlled, and are in static equilibrium with the external forces. These stresses are strictly limited to a value less than the material yield stress, defined by a design allowable, Sm. Sm is approximately 2/3 of the temperature dependent yield strength.
>
> *Secondary* (Q) stresses are general thermal or geometric discontinuity stresses which are strain limited. Only their range is limited to the yield range of 3Sm.
>
> *Peak* (F) stresses are highly constrained local peaks caused, for instance, by small notches or local thermal hot spots which cause no nett deformation. It is common to isolate F stresses by linearizing the stresses on a cross-section. The F, or peak stress, is the deviation from this

statically equivalent linear distributions. There are problems in deciding how to define the F stress in 3-dimensional components where sections cannot be so easily identified.

The ASME stress categorization scheme is a reminder that all stresses are not equal, but that their implications in terms of overall structural failure must be taken into account.

The mandated nature of stress limits is an important feature of the ASME Code, and this carries over into N47. The ASME stress categories are the result of a great deal of careful thought. Consequently, they have been adopted virtually unchanged throughout the field of pressure vessel design and, by extension, throughout the related field of high temperature design.

Special High Temperature Failure Mechanisms

N47 considers a number of temperature specific failure mechanisms which are additional to the low temperature cases required by the main ASME Code.

These are,

i) Steady creep deformation

ii) Creep ratchetting, or magnified creep deformation due to load cycling and creep/plasticity interaction

iii) Creep/fatigue interaction.

Steady operation is dealt with in N47 by replacing Sm with a time dependent limit, St, calculated from creep deformation or rupture date for the assumed design life of the component. The actual calculation of these limits is carefully regulated, and consequently rather involved. For the purpose of discussion, St can be considered as a lower bound on the creep rupture strength at, say, 250,000 hours.

The current version of N47 deals with creep ratchetting using a modification of the Bree Diagram, incorporating an upper bound to creep using the O'Donnel/Porowsky "elastic core" concept.

The method is adequate for its designated purpose, which is the problem of steady

Pm stress on a plate combined with a cyclic thermal bending Q stress. This is extremely limited in scope and forms one of the shortcomings of N47. A question which still has to be answered is, how to deal with more general cases of combined mechanical/thermal cyclic deformation Cox and Ponter [2] has addressed this issue in a recent European Community Report.

One aspect of N47 that has drawn a great deal of debate is its approach to creep/fatigue evaluation. The combined effects of creep and fatigue are assessed by calculating fatigue and creep damage separately for the cycle, using a Robinson life fraction rule for creep and Miner's Rule for fatigue. Design creep and fatigue curves are supplied in the Code Case. Finally the two damages are added according to a Linear Damage Summation Rule (Fig. 2).

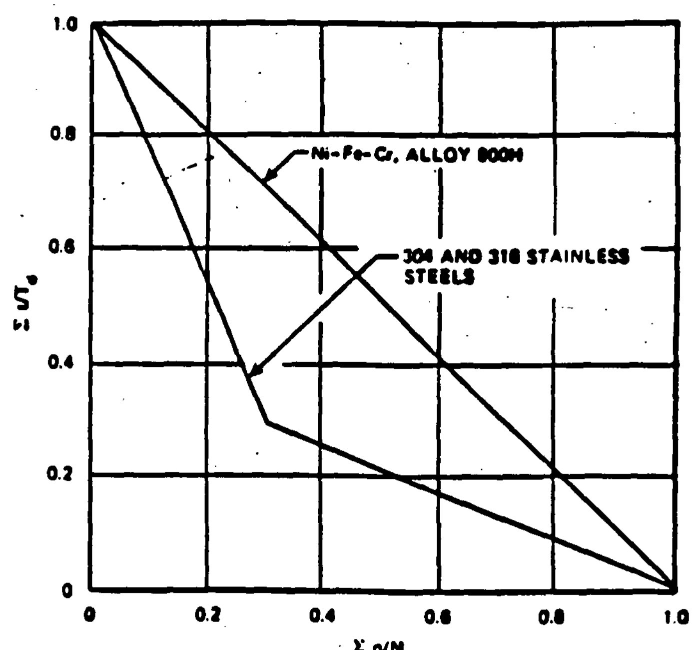

Figure 2. The N47 Linear Damage Summation Rule

Alternative Elastic and Inelastic Procedures for Creep/Fatigue

N47 originated at a time when detailed methods of analysis like FEA were not as

widely used as they are today. It therefore offers two alternative methods for evaluating creep/fatigue damage, an approximate "elastic" route, and a more complex inelastic route. The latter is described very briefly in N47, since it assumes that precise stress/strain histories at critical points in the components can be found, and will not be discussed further here.

The elastic route has found wide application, and much of the Code Case developers' efforts have gone into it. The only fundamental difference between this and the inelastic route is that approximate methods are prescribed in N47 to estimate stress/strain cycles at critical locations.

The elastic route also uses a different set of high temperature fatigue curves from those adopted for the inelastic route. These curves, the infamous Fig. T-1430, incorporate some time-dependent creep damage accumulated during hold times. The result is that creep damage is counted twice in calculating combined creep/fatigue damage. The only rationale for this procedure seems to be that the Code Case developers, recognizing the confused state in the field of creep/fatigue interaction, took an understandably conservative approach and included creep damage wherever it might be appropriate. As understanding has improved N47 has been revised, and the latest revisions include eliminating Fig. T-1430, and unifying the damage evaluation procedure for both elastic and inelastic routes.

N47 uses isochronous curves exclusively for representing creep deformation. This is an interesting development because the preferred approach, from the applied mechanics point of view was, and still is, to represent constitutive behavior in rate form. In practical terms, however, isochronous curves have some advantages which will be discussed later. One intriguing advantage is that analysis using isochronous curves is quite accurate with considerably less computational effort than might be needed by more sophisticated techniques.

<u>Shortcomings of N47</u>

i) N47 takes no account of defects.

N47 has its beginning before time-independent, inelastic fracture

mechanics had been developed as an engineering tool. None of the work of recent years on creep crack initiation and growth is therefore addressed in the Code Case.

ii) Limited Scope of Creep Ratchetting Analysis

While having stood the test of time for problems within its scope, that scope is admittedly limited, and there is a need for more general procedures to cope with complex geometries in a systematic manner.

2.2 Recent Progress in Design Guideline Development

Several countries, notably the UK and France, have developed recently high temperature design strategies of their own. These have been reviewed comprehensively in a recent SMIRT Post-Conference Seminar, and will not be discussed here. Most of these developments do not differ greatly in concept from N47. They tend to adhere to variations of the Bree Diagram to deal with ratchetting, continue to use a life fraction role for creep under varying conditions, and a Linear Damage Summation Rule for creep/fatigue interaction.

The most radical exceptions to this rule are the new British "R5" (3) procedure and the French RCC-MR Rules [4] for Fast Reactor Design.

It is interesting that, after more than 20 years of intensive research into high temperature material behavior, improvements or changes in design procedures derived from this newfound knowledge are, on the whole, rather marginal when compared with the longstanding N47 approach. This is, of course, a general statement, and islands of significantly improved understanding can be found in specific areas.

Creep/fatigue interaction in particular has been the focus of a great deal of research work since the early 1970's. The result has been a proliferation of competing theories ranging from the purely empirical through to mechanistic model based on detailed microstructural observations, and everything in between. Of these, the best known are the

original Linear Damage Summation Rule, used in N47, Strain Range Partitioning, and Coffin's Frequency Modification. There are many other contenders which have been reviewed many times, e.g., by Viswanathan [5] will not be mentioned any further here.

As of the beginning of the 1990's it appears that the earliest and simplest concepts continue to hold up. There does not seem to be a good reason to use anything more complicated than Linear Damage Summation using a simple Robinson-type life fraction rule for continuum creep damage, and Miner's law for fatigue damage. This conclusion is widely reflected in recent design code proposals.

Recent work in the UK, for instance, suggests that there is very little in the way of true creep/fatigue interaction. In some materials, such as austenitic stainless steel, for instance, there is a perceptible rate effect on fatigue life, which can be traced to rate-dependent variations in yield stress and strain hardening index, but this is only active for hold times in the region of a minute to an hour. Other presumed creep/fatigue interaction effects can be traced to fatigue/oxidation effects.

The lesson seems to be that simple damage models are sufficient provided the operating cycle can be predicted with reasonable accuracy. Thereafter, the question of what constitutes the "right" damage model for a given component of damage is of secondary importance. It is more important to choose a model that is capable of being used at all, given the limited data generally available at the design state. Examples in practice which support the procedures adopted by the Codes are given below.

Reformer tubes are case Nickel alloy tubes used in the chemical industry. They are pressurized with steam at high pressure and high temperatures approaching 900C. These tubes have a poor record of premature failure by creep rupture. Using simple damage concepts, Jaske [6] has been able to show that the disparity between predicted and service performance in these parts is largely due to a failure to take realistic operating conditions into account in design. The original design was based on a steady state conditions whereas the actual operating conditions include large amounts of relaxation creep during the brief startup transients. There is a big difference (approaching a factor of 5) between the short term yield strength and the creep strength of the material at these temperatures, which leads

to exceptionally large amounts of creep relaxation in a short time. This, incidentally is typical of very high temperature operation generally.

The work of Neu and Sehitoglu [7] and his students on thermo-mechanical fatigue is further support for the proposition that relatively simple damage models carefully chosen and carefully can deal with most problems of damage accumulations. Sehitoglu has been able to explain all the essential features of thermo-mechanical fatigue by realistically modelling the stress/strain/temperature history of a load cycle and evaluating the separate components of material damage due to fatigue, creep and surface oxidation, with simple, well established models.

Two areas where more recent work can provide significant improvements over N47 are Creep Ratchetting and Creep Cracking. The British R5 approach deals with both of these questions in the framework of an new integrated approach to high temperature design. Creep ratchetting is also dealt with in a novel way by the French RCC-MR Rules. These two new developments will now be discussed as far as they contribute fresh thinking to the high temperature design process.

2.3 The French RCC-MR Rules (4)

The RCC-MR rules were developed specifically for the French FBR program. They therefore concentrate on typical metal cooled reactor operation, i.e., low primary stresses and large, rapid thermal gradients. This puts the guideline in close company with N47, which was also developed largely with FBR design in mind.

As far as primary design is concerned FCC-MR does not differ significantly from N47. There are differences in the way design quantities such as Sm and St are calculated from material test data, but these are mostly fine tuning. In fact, the ASME notation of P, Q and F stresses has been carried over almost intact.

The same is true of fatigue and creep damage assessment. RCC-MR shows no significant departure from the basic N47 procedure of Linear Damage Summation, although there are differences in detail on how to calculate the individual components of damage.

The numerical differences between FCC-MR and N47 are about evenly distributed either way. To a third party observer, however, the most striking feature of the two set of rules, N47 and FCC-MR, is how closely they parallel each other, to the extent that they have been tabulated in one German study and compared virtually rule-by-rule.

The main difference between FCC-MR and N47 lies in the former's much more comprehensive treatment of creep ratchetting. The architect of this alternative approach is Roche [8]. He argues that the essentials of cyclic deformation in the creep range is too difficult to capture more than qualitatively by purely theoretical models. He proposes a simplified method using an experimentally derived interaction curve based on tests with real complex structural shapes.

There may be a strong point to this argument, especially when dealing with materials similar to austenitic stainless steel, which strain harden heavily, and differently under motonic, in-phase cyclic and out-of-phase cyclic loading.

The French approach to ratchetting, although more comprehensive than N47, is still not inclusive. It deals only with the Bree type of problem. This is what Cocks and Ponter refer to as a Class A [2] shakedown problem. Two other classes of shakedown problem, are also likely to be experienced in fast reactors, are not considered in RCC-MR.

2.4 The R5 Procedures (3)

The development of R5 was begun by the one time Central Electricity Generating Board (CEGB), and completed by the same organization in its "privatized" guise of Nuclear Electric Plc.

In addition to original work by members of that organization, R5 draws on many concepts developed in Cambridge, Leicester and Liverpool Universities in the 1960's and 1970's. The most ubiquitous of these is the Reference Stress.

The Reference Stress

The Reference Stress concept was developed in the 1960's as a pragmatic tool for finding approximate estimates of deformations, and later creep rupture times, for complex

components, in the absence of either a method for exact inelastic analysis of the component itself, or a full description of the material. Both of these problems were endemic in high temperature design at the time, but were believed to be a temporary state of affairs which would be resolved as more research was done.

In essence the Reference Stress concept allows overall creep deformations and ductile creep rupture of complex components to be predicted with as little information as an estimate of the component limit load, a single creep test at a carefully chosen stress level called the "Reference Stress".

Although more detailed analyses of creep behavior are now possible, the Reference Stress concept still has value even today. The method does not give a detailed point-to-point description of creep deformation, but is surprisingly accurate for those few quantities it sets out to predict. For many practical purposes these quantities are all that is needed to make sensible engineering decisions. Furthermore, a detailed analysis of complex components, <u>at the stage where they are actively being designed</u>, is often a logical impossibility, given the data available under the constraints of limited and <u>irreversible</u> time that typically exist during design

Steady Stress Deformation and Rupture

The Reference Stress has gained a new life as a practical means of condensing a great deal of complex material/component interaction into a manageable form. This, rather than the ASME Primary stress categorism, is used in R5 as a measure of primary stress for evaluating long term, pseudo-steady creep deformation and rupture criteria. The Reference stress adopted in R5 is a simplified version developed by Sim.

$$\sigma_{ref} = \frac{P}{P_L}\sigma_y \leq \sigma_T$$

where σ_y, σ_T etc. have the usual ASME CC N47 meanings σ_T can be the stress-to-rupture in a specified time, or the stress-to-1%-strain, for instance, whatever the creep-based

failure criterion may be.

The ASME stress classifications are retained in R5, at least for the purpose of discussing the main components of stress. For example, a general thermal stress will be referred to in R5 as a "thermal Q" stress. This does not conflict with the use of the Reference Stress concept. In fact, the Reference Stress is a very natural way of generalizing Primary Stress components to complex geometries where simple linearizing techniques no longer work.

Creep Ratchetting and Shakedown

The R5 treatment of shakedown is a radical departure from previous practice as epitomized in N47. The basic approach is a return to the concepts of Melan and Koiter [ ]. Shakedown is assured by finding a residual stress state which, when added to the applied mechanical stresses, produces a cyclic stress state, $\sigma(t)$ which nowhere exceeds the yield condition. This basic concept is generalized to the time dependent state by using principles described by Goodall et al [9]

Shakedown is achieved if the sum of two elastic stress distributions, $\sigma(t)$, the linear elastic time varying stress history, and ρ, a constant residual stress, satisfy the condition

$$f(\sigma(t)+\rho) \leq K\sigma_y$$

where f(..) is the yield condition, and K is an experimentally determined factor ensuring that <u>material</u> ratchetting does not occur below a stress level of $K\sigma_y$.

Once this cyclic state has been determined R5 provides a systematic procedure for evaluating time dependent deformations based on a bounding theorem which will not be discussed here.

The stresses $\sigma(t)$ and ρ ignore peak, or F stresses, and only include what N47 would call the (Pm+Pb+Q) stresses. For practical purposes, $\sigma(t)$ is the time varying (Pm+Pb+Q) stress history, and the residual stress, ρ, can be chosen to be proportional to the thermal stress range, ΔQ. Both of these stress distributions are routine design office

calculations. ANSYS, for instance, is almost standard equipment in design offices today, and this program has a postprocessing procedure to extract the required stress components automatically.

Although the derivation of the principles involved in this cyclic analysis is complex, the method itself is relatively simple to implement, especially today, when linear elastic analysis of extremely complex geometries has become a standard design office procedure using FEA packages like ANSYS or ABAQUS.

Creep Damage

"Ductility exhaustion", or a strain fraction rule is accepted in some cases by R5 as an alternative to the more common life fraction rule for creep damage accumulation. The concept is identical in principle to the life fraction rule except that the fraction of creep ductility exhaustion at a given stress level is assumed to be the damage parameter.

Creep Cracking

R5 is currently unique among systematic design guidelines in providing an explicit procedure for evaluating defect tolerance in the creep range, i.e., evaluation of creep cracking.

Creep cracking is a phenomenon which has been known for many years, and has been known to correlate reasonably well with a time dependent version of the post yield fracture parameter, J. The time dependent quantity is referred to as C. In general J can be written as

$$J = A\sigma \in \pi a$$

This is a robust concept which carries over into the creep range by the simple expedient of replacing "strain" by "creep strain rate". Hence

$$C = A\sigma \dot{\in} \pi a$$

Practical evaluation of creep cracking has been hampered in the past by the complex methods proposed for dealing with it. R5 makes a real contribution by adopting a simple procedure for computing C developed by Ainsworth and his colleagues, which removes this difficulty. The expression for C is given by

$$C = R\sigma_{ref}\,\dot{\epsilon}_{ref}$$

where σ_{ref} is the reference stress for the cracked section, $\dot{\epsilon}_{ref}$ is the corresponding creep strain rate, and R is a geometry and load-dependent size parameter obtained from the linear elastic stress intensity for the same configuration by the relation

$$R = \left(\frac{K}{\sigma_{ref}}\right)^2$$

where K is the linear elastic stress intensity for the geometry and load cofiguration under consideration.

The approximation has been shown to be accurate within about 10 to 15%, which is satisfactory in most practical circumstances.

R5 does not only consider creep crack growth. The defect assessment process is a comprehensive one. It also considers the incubation time to initiate a creep crack in service the prior creep continuum damage, and the time to initiate creep crack propagation in a preexisting crack.

Shortcomings of R5

The only obvious shortcoming in R5 appears to be that it cannot be carried out entirely using linear elastic analysis. At some point Reference Stresses have to be calculated, which require an estimate of the limit load of the component. Many limit solutions for standard geometries can be found in the open literature, however, and R5 gets

around the problem by making use of them, as did its low temperature predecessor R6. There are also ways of getting around this problem by creative use of linear elastic FE programs, so that the problem is not a deficiency.

2.5 Summary of Progress in High Temperature Design

The procedures described in N47, R5 and RCC-MR deal with "conventional" high temperature design, i.e., applications well within the bounds of current experience, using well known materials with properties that are both qualitatively and quantitatively understood. It is believed that this is an area which can now be dealt with reasonably competently.

Most importantly there has been a great deal of convergence and agreement among these, and other, high temperature design procedures on basic issues such as the nature of damage, and how to calculate it. The procedures themselves do not differ substantially from those being considered 20 years ago, but a consensus is emerging which confirms the validity of the original concepts. Remaining variations do exist in creep ratchetting but even this problem is apparently resolved.

Any one of the reviewed procedures can be expected to cope satisfactorily with deformation and continuum damage. So far only R5 is adequately equipped to deal with cracks and defect tolerance. If one procedure has to be adopted in its entirety, R5 would seem to be the best choice at the present time.

The lesson seems to be that simple concepts of damage evaluation are adequate as long as a good job is done of modelling the service history and predicting the component response to it. This is encouraging, because it indicates that the most productive design efforts are those put into realistic modelling of transient component modelling. which is more readily achieved as a result of the increasingly widespread availability of FEA packages running on workstations.

Problem Remaining

The optimistic tone expressed above notwithstanding, work remains to be done in the general area of high temperature design, particularly at very high temperatures. Problems for which existing design procedures currently have no satisfactory answers include:

i) Long term material behavior including yield stress which cannot be characterized in anything approaching exact terms.

ii) The identification of F stresses in complex 3-dimensional components where the ASME linearization process breaks down.

iii) Use of brittle materials such as intermetallics or ceramics for very high temperature applications.

To use materials with changing properties, it may be necessary to abandon one of the foundations of current design code philosophy. This is the fixed and non-negotiable yield strength and the design criteria based on it.

For instance, how is Sm determined for a material like bainitic Cr/Mo pressure vessel steel which can cyclically soften by as much as 30%, but only in localized areas? The initial yield strength is too optimistic. On the other hand it is needlessly conservative to adopt the fully softened state for design purposes because there may be applications which do not experience cyclic loading at all. Does one use different values of Sm depending on the expected load history? This simple question requires some attention on the part of code developers.

A more complex problem is posed by a structure where cyclic softening only occurs in localized areas of constrained deformation. More definite rules are needed to determine when an area of local high strain range is truly constrained and when it becomes unconstrained. If it can be shown to be constrained then local aging effects are not critical. If it cannot, it may be necessary to use the aged or softened condition of the material in design. An attempt to resolve this problem has been given by Marriott and Handrock [9, 10] who suggest methods for describing complex material behavior in terms suitable for use in the High Temperature Codes.

Strain rate sensitivity is only one more example of complex constitutive behavior encountered at very high temperatures. What should be taken for normally well defined properties like the yield stress when this quantity is not remotely constant? The problem can be dealt with artificially by specifying an arbitrary strain rate for tensile testing, but this does not answer the question of how to deal with problems like thermal transients, in which strain rates vary a great deal, but not slowly enough to be considered as creep.

Methods like Reference Stress techniques, which rely on an adequate, although often unspecified, amount of reserve ductility to redistribute stresses, cannot be assumed to apply to the new breed of semi-brittle materials. New design concepts are needed to deal with such materials.

2.6 Conclusions

i. For the most part, the net result of the work done on high temperature design over the past two decades is that there are no surprises. Long standing concepts, particularly the simple ones about damage characterization, have stood the test of time very well.

ii. The most productive strategy for reliability in future design is to retain the familiar, long standing concepts, but to determine an accurate representation of realistic service conditions.

iii. For mature high temperature applications, any one of the three existing design guidelines, N47, R5 and RCC-MR appears to be equally acceptable within their individual scopes.

iv. R5 is the most comprehensive and integrated of the three contenders. However, it has not had the extensive field testing enjoyed by N47, so it needs to be used with caution until confidence is developed.

v. Very high temperature applications, in which materials are pushed to the very limits of their capabilities, pose some special problems for future developers of design codes and guidelines. There will be a need for more flexible thinking about design allowables such as yield strengths, which have been assumed in the past to be fixed quantities. Strain rate effects and service related aging are examples of complexities that need to be

accommodated.

vi. Some thought needs to be given to the question of design in semi-brittle materials such as intermetallics, which have marginal amounts of ductility, sufficient to accommodate peak (F) stresses, but little more.

Detailed recommendations for exploiting and extending the Codes are given in Appendix A "Framework for High Temperature Design Procedures."

3. THE FREEDOM SOLAR POWER SYSTEM

3.1 System Description

The elements of the system are shown schematically in Fig.1. The canisters (1) contain the eutectic material which has a melting temperature of 764°C (1416°F) and absorbs the energy of the sun by melting while supplying heat at constant temperatures by freezing during the dark period. The cycle time is 90 min so that in a life of 30 years, the eutectic will experience 175000 cycles of melting and freezing.

The heat exchanger tubes (2) are 8 ft. long with 0.875" O.D and thickness 0.035". Each tube supports 96 containment cannisters along the length and they in turn are attached to an annular manifold which maintains the 82 tubes in a cylindrical geometry and which supplies and collects the working fluid at a maximum pressure of 73.0 lb/in^2 (508 kPa) and temperatures of 760°C (1400°F).

The inlet and outlet manifolds (3) and (4) supply the working fluid to the heat exchanger at 649°C (1200°F) and collect it at 760°C (1400°F) after absorbing heat from the cannisters.

The inlet and outlet pipes (5) and (6) are attached to the support structure by bellows so that the displacement induced by thermal mismatch can be accommodated.

3.2 Steady State Operating Condition

The working fluid pressure varies between 50 (345 kPa) and 75 (508 kPa) lb/in^2 while the temperature varies between 1200^oF (649^oC) at inlet to the heat exchanger and 1400^oF (760^oC) at the outlet. The temperature of the tube is approximately 200^oF (93^oC) higher than the working fluid but never exceeds 1400^oF (760^oC). The initial temperatures appear to occur in the cannisters.

3.3 Material Properties (HAYNES 188)

The material selected is a cobalt based alloy (HAYNES 188). Considerable data exist for this material with creep data obtained from tests of 20,000 hr duration. It is a common problem that even 20,000 hr tests are of short duration compared to the design life term of 260,000 hr and extrapolation techniques are required to estimate long-term material properties.

Unfortunately an understanding of the aging processes are not readily available. It is possible to predict creep data using the Larsson-Miller parameter, but since this approach is not mechanism based the predictions must be used with caution. The collection of creep data of Frost and Ashby [11] contains very little information on cobalt but the report does give some general results for hexagonal metals. Furthermore, a method called Normalization of Constitutive Laws purports to normalize the properties for systems with identical crystalline structure so that a series of master curves can be produced. This procedure has not been attempted in this report but it would appear that the suggested procedures offer another method for establishing long-term data which is mechanism based. Using the date available and using Larsson-Miller extrapolation techniques the following data are tentatively suggested

Temp ^oF (^oC)	1200 (1182)	1400 (760)	1520 (827)	1700 (927)
UTS ksi (MPa)		75 (517)		36 (248)
0.2% σ_y ksi (MPa)			43 (296)	
20 (138)				
Ductility %		35		40
Rupture Stress ksi (MPa)		10 (69)	4.2 (29)	
Stress for 1% strain ksi (MPa)		5.5 (38)	2.5 (17)	
E ksi (aPa)		$24{\cdot}6 \times 10^6$(169)	$24{\cdot}6 \times 10^6$(169)	
$\alpha/^oF$ (^oC)		8.74×10^{-6} (15.7×10^{-6})	8.9×10^{-6} (16.0×10^{-6})	

Properties drop off rapidly at temperatures greater than 1600^{o}F (871^{o}C)

TABLE 1 Tentative Prediction of Properties of HAYNES 188

The data in the above table is sufficient to perform all the existing code calculations with the exception of fatigue, creep-fatigue and defect analysis.

The fatigue strength can be obtained from the general fatigue curve for 1400^{o}F (760^{o}C)

$$\Delta\varepsilon = \frac{0.0138}{(N_f)^{0.125}} + \frac{0.496}{(N_f)^{0.701}}$$

where $\Delta\varepsilon$ is the total cyclic strain range.

For $N_f = 17500$ this result gives

$$\Delta\varepsilon = 0.31\%$$

which corresponds to an elastic stress of 75 ksi (513 MPa). It is unlikely therefore that pure fatigue is likely to be a problem.

The creep damage accumulated during stress relaxation could be an important factor in determining life in creep-fatigue. Relaxation data at is only available at high stress levels and is insufficient for satisfactory predictions of the life of components under creep-fatigue conditions likely to occur in practice. A major deficiency is the lack of information about long term creep rupture strength which is currently estimated using the Larsson-Miller extrapolation.

No information could be extracted about the toughness of the material. In the absence of this information an approximate estimate has been proposed by Ritchie [ ] with

$$\frac{K_{Ic}{}^2}{E} = \Pi \sigma_y \varepsilon_p L$$

where L is a characteristic length which is taken to be 0.1 mm and ε_p is the ductility. With

$\sigma_y = 517$ MPa $\quad \varepsilon_p = 0.35 \quad L = .1$ mm $\quad E = 169$ MPa

$$K_{ic} = 98\text{MPa}\sqrt{m}$$

With $\varepsilon_p = 0.35$ this value would correspond to plane stress conditions. For plane strain conditions the failure strain might be lower by a factor of 5 so that the toughness is estimated to be

$$K_{ic} = 20\text{MPa}\sqrt{m}$$

This cannot be regarded as a reliable value and is probably a lower bound. However, what is important is that the toughness will drop at cryogenic temperatures below the Transition Temperature when K_{ic} could drop to say $5\text{MPa}\sqrt{m}$. Clearly toughness measurements are required.

4. IMPLICATIONS OF CODES ON THE DESIGN OF THE SOLAR PANEL

4.1 Introduction

In this section some observations are made about the system which have been stimulated by the study of the existing Codes. No attempt is made to perform a detailed study but some issues arise naturally from the systematic application of the Codes. These are now discussed.

4.2 Definition of Loads and Stress Levels

The Codes define loading conditions as

i) Normal Operation (100% of time)

ii) Normal Transients (Many)

iii) Upset Conditions (Few)

iv) Accident (Unlikely events but must be considered)

In the case of the Freedom Solar Panel the loading for Normal Operation (i) is the working fluid pressure which varies between 50 and 73 lb/in^2 [345 and 508 kPa] and results in a circumstantial stress in the tube of 6.35 ksi (43.5 MPa). The Normal Transient Loading (iii) is thermally induced. An approximate analysis of the cannisters suggests that the most critically stressed component are the cannisters with a bending stress of 2.2 ksi (15 MPa) due to longitudinal thermal mismatch. In the radial membrane the stresses are -3.3 ksi (23MPa) in the circumferential direction and 0.6 Ksi (9 MPa) in the radial direction. These stresses appear to be small but should be checked by a full elastic analysis. In addition to the above loading the effect of asymmetric heating of the tube bundles will result in differential expansion in the longitudinal loading with a consequent loading of the manifold in the direction of the tubes. Asymmetric canister heating causing differential heating around individual tubes will induce curvature in the tubes thereby applying moments at the intersection of tube and the manifold.

The primary stress corresponding to Normal Operation resulting from the working pressure is 43.5 MPa. Including the usual factor of safety of 1.5 the primary stress

corresponding to the Design Load is 65 MPa. Referring to Table 1 the stress which causes 1% creep strain in 30 years is estimated to be 38 MPa and on this bases the design is inadequate. However, the 1% criterion of the Codes has been proposed for large structures and is probably excessively severe [since a 1% in the tube corresponds to a radius increase of 4.4×10^{-3} in or 0.1 mm]. The rupture stress for 30 years from Table 1 is 69 MPa so that according to this estimate there would be no rupture. This small calculation does emphasize two points however. First, the need to estimate the long term creep rupture strength of the material and secondly, to ease the 1% life time strain criterion used in the existing codes to perhaps a 5% strain criterion.

The Upset Condition (iii) is likely to correspond to a misalignment condition when the sun rays are not captured or when the system is temporarily shut down. Then the effect of the start-up conditions and the stress fields induced should be determined. An estimate of the thermal stress is $E\alpha T$ and with the values in Table 1 with T = (273+760) the stress upset is 40 ksi (275 MPa). According to the Codes the allowable stress for this type of thermal loading is twice the yield stress or 385 MPa in so that the Code criterion can possibly met during start-up. It is emphasized, however, that a thorough analysis of start-up conditions in justified, and it may be that high thermal stresses must be reduced by appropriate design solutions.

The Accident Condition (iv) is likely to arise when the cannisters fail. Then local disturbance of the temperature field shall occur and attempts should be made to determine the stresses. Again, the Codes allow higher stress levels for this type of loading which is rare and thermally induced.

4.3 Defect Analysis

Defects in the form of cracks can occur in welds and their study is required in the Code R5.

The primary working stress in the tubes is 6.35 ksi (43.5 MPa). Assuming a value of Kic = $20 \text{ksi}\sqrt{\text{in}}$ (estimate) the critical crack length a is given by

$$a = \frac{1}{\Pi}\left(\frac{K_{ic}}{\sigma}\right)^2 = \frac{1}{\Pi}\left(\frac{20}{6.35}\right)^2 = 3.2\text{in}$$

which is much greater than the wall thickness of 0.035 in. Hence, it would be reasonable to deduce that leakage would certainly occur before general fracture. The effect of working below the Transition Temperature needs to be considered.

Because the primary stresses are dictated by long term conditions the primary stresses are small and the value of the applied stress intensity is small. For example, if a through crack exists which is 3/4 of the tube thickness then the value of K is given by

$$K = 0.3\sigma\sqrt{4\Pi}\sqrt{t}$$

with $\sigma = 6.35$ ksi and $t = 0.035$ in the value of K is

$$K = 1.26\text{ksi}\sqrt{\text{in}}$$

which is very small and certainly smaller than any reasonable value of K_{ic} $20\text{ksi}\sqrt{\text{in}}$ (estimated).

Normally in R5 attempts are made to estimate the crack growth rate under fatigue and creep conditions. The required data is not available but an attempt has been made to estimate the importance of creep/fatigue crack growth in Appendix B, Section 1.5.4. It is suggested by these calculations that creep/fatigue crack growth deserves closer attention and that of crack growth data would be desirable.

4.4 Fatigue and Creep-Fatigue Analysis

In Section 3.3 an analyses of fatigue suggests that this problem does not exist. Attempts in Appendix B, Section 2.3 to perform a creep/fatigue analysis were threatened by lack of relaxation data. The data available is associated with stress levels substantially higher than that occurring in practice and attempts to establish extrapolation techniques

were unsuccessful.

5. CONCLUSIONS AND RECOMMENDATIONS

1. The Codes which are available for high temperature design collectively provide a sound base for the design of space power plant with lives of many years. Any one of the reviewed procedures can be expected to cope satisfactorily with deformation and continuum damage. So far only R5 is adequately equipped to deal with crack and defect tolerance. If one procedure has to be adopted in its entirety, R5 would seem to be the best choice at the present time.

2. A dominant feature of the Codes is the definition of loading types and the corresponding allowable stress levels. It is important, therefore, to determine the loading classifications and magnitudes.

3. The tentative and simple calculations performed in the report suggest that designs are likely to be dictated by the long term creep characteristics of the material. These material properties can presently only be estimated by extrapolation techniques which are untried for the cobalt alloys proposed. It is recommended, however, that extrapolation techniques including the normalization technique of Frost and Ashby [11] be investigated so that use can be made of the extensive data available for other hexagonal metals

Appendix A A Proposed Draft for High Temperature Design Procedure

Summary of basic design procedure.

A1 Material Property Definitions

A1.1 Tensile property data vs operating temperature.

Recommend using the same rules as adopted by ASME BPV Code.

i.e. Time independent allowable stress, $Sm = Min (UTS/2.35, \sigma_y/1.5)$

Time dependent allowable stress, St = Min (Stress for 1% strain in 10^5 hours, or rupture in $3x10^5$ hours).

Primary Allowable Stress Intensity $Smt = Min (Sm, St)$

A1.2 Creep Material Data

i) Creep Deformation

The conventional method of representing creep data in basic material studies is in the form of strain/time curves at constant nominal stress. It has been found that, for the purposes of design computation, it is more convenient to cross-plot creep data in the form of so-called *isochronous stress-strain curves.* It is possible to present data more compactly this way. It also avoids the complication of distinguishing between initial plastic deformation and the rapid, time dependent creep deformations experienced during the early stages of primary creep. All the design codes and guidelines discussed here use this format for material presentation, as well as utilizing it directly in various types of component computation.

ii) Creep Rupture Data

The conventional method of presentation for this information in all codes and guidelines is either as log(σr) vs log(tr) curves, or in the form of a Larson-Miller plot. The latter is the most common method of extrapolating from short-term, high temperature test data to produce long-term design data, despite the existence of many other, more sophisticated extrapolation techniques.

A1.3. Continuum Fatigue Data (No hold time)

Basic Material data:

Compile all available short time fatigue data on HAYNES 188 into bands for specific temperatures into single plots, and establish mean experimental data.

Design Curves:

Design curve - Min (Basic Data Curve, stress/2, Nf/20)

A1.4. Creep/Fatigue Interaction

Calculate creep and fatigue damage separately, then use bi-linear damage summation to combine damages.

For materials with minimal experimental data, use ASME interaction diagram for austenitic steels, with Dc = Df = 0.3. This is probably conservative, but satisfactory for initial design calculations.

A2 Definition of Load Conditions

The following load/thermal conditions need to be considered in the design of any component intended for high temperature operation.

Note that component loadings consist of two distinct types.

The first category is *PRIMARY LOAD*. Primary loading is caused by external mechanical loads. The stresses due to primary loads are not relieved by relaxation.

The second category if *SECONDARY LOADING*. Secondary loads are self limiting, such as thermal distortion, and cannot, of themselves, cause general yielding

collapse of a component. Secondary stresses are relieved by relaxation.

A2.1. Normal Operation

Definition: Static conditions experienced regularly during service life.

Cyclic conditions occurring often enough to involve significant fatigue risk by themselves.

Loads to consider:

1. All Static pressure/temperature worst case combinations.
2. System loadings, i.e. external primary loadings on components due to supports and component-to-component thermal loadings which need to be considered primary because of the prospect of elastic followup.

   e.g nozzle loads on solar collector manifolds, due to differential thermal distortion of tubing.
3. Normal operational thermo/mechanical cycles, e.g. 90 minute orbit.
4. Regular startup-shutdown cycles, if any, e.g. if more than several/year.
5. Frequently anticipated system trips.

Design Criteria:
1. Limit Load Collapse
2. Creep deformation and rupture limitation.
3. Incremental Collapse (including Creep Ratchetting).
4. Creep/fatigue evaluation.

A2.2 Upset Conditions

Definition: Fault conditions or deviations from normal operation which are expected in service with a frequency greater than 1, not sufficient to cause creep/fatigue failure in themselves but a possible significant contribution to damage incurred under normal operations.

Loads to consider:

1. Preservice test loadings, if any, e.g. hydrotest.

2. Infrequent system trips.

Design Criteria: 1. Limit Load Collapse

2. Incremental Creep/Fatigue Damage per event.

A2.3. Accident Conditions

Definition: Load states which are unlikely (i.e. probability << 1), but physically feasible.

Loads to Consider:

1. Rare event external loadings, e.g. collisions, overpressures due to system failures.
2. Secondary effects of non-catastrophic primary failures, e.g. thermal fatigue failure of primary circuit tubing due to "coldspot" caused by loss of eutectic from a canister in solar collector.

Design Criteria: 1. Limit Load Analysis

2. Creep/fatigue analysis of secondary effects.

A2.4. Defect Tolerance

Definition: Repeat of Normal, Upset and Accident analyses for critical points, including presence of a hypothetical worst case initial defect.

Loads to Consider:

1. Steady load creep cracking of worst case defect under Normal and Upset conditions.
2. Creep/fatigue crack growth under Normal and Upset conditions.
3. Ductile tearing and/or brittle fracture under Normal, Upset and Accident conditions.

Design Criteria: 1. Creep Crack growth rate, da/dt vs K or C* (very long dwells)

2. Creep/fatigue crack growth, da/dN vs ΔK or ΔJ, including frequency effects.

3. Fracture resistance to single large overload, applied load vs K_{1c} or J_R curve.

A3. Design Procedure - Background

This procedure follows the basic format adopted more or less uniformly by the ASME BPV Code Case N47, the French high temperature design standard RCC-MP for metal cooled fast reactor construction and the British "R5" design guidelines, originally developed by the (then) CEGB. The Japanese use essentially the ASME approach, with minor modifications to suit their specific materials, while the European countries, other than France and the UK have their own standards which, again, appear to be relatively minor deviations from the French RCC-MP standard, to accommodate unique national code regulations.

The version presented below closely follows the recommendations of the SI Working Group of the UKAEA for the design guideline for fast breeder reactor component in the UK. This guideline draws freely from RCC-MP for the majority of basic design procedures and design limits, but uses the British "R5" guidelines as the prototype for its method of dealing with incremental collapse and defect tolerance.

A4. The Structure of High Temperature Design

A4.1. Outline of the Basic Stages in the Design Process

There is a structure of five basic stages common to all the design procedures quoted above. R5 and the UKAEA assessment procedure add a 6th stage dealing with defect tolerance. Procedures for this final step are taken almost exclusively from the R5 guideline.

I. Primary Limit Load Assessment - Resistance to Collapse under a single steady load application.

II. Shakedown Assessment - Resistance to incremental collapse under multiple load applications.

III. Creep Damage Assessment.

IV. Fatigue Damage Assessment.

V. Creep/Fatigue Interaction.

VI. Defect Tolerance.

Stages I and II are considered mandatory precursors to any subsequent analysis of damage accumulation.

Stage I, Primary Load Assessment, ensures that the component is capable of withstanding at least one extreme load application without suffering excess deformation of collapse by a general yielding mechanisms. This applies to both short term loads and excess creep deformations under sustained steady load. Only mechanical loads are considered in Stage I, because thermal and other secondary loadings cannot produce collapse.

Stage II, Shakedown Assessment, is concerned with the ability of the component to maintain dimensional stability under a succession of mechanical and thermal transients. Stage II covers two requirements. The first is the absolute dimensional stability under short term, or low temperature load transients, i.e. no plastic ratchetting. The second is that there should be no accelerated creep deformation as a result of interaction with cyclically induced plastic deformations caused by the transients.

If the requirements of both stages I and II are met, it can then be claimed that any significant inelastic deformations on the time scale of a typical transient (e.g. minutes, hours or days) will be entirely *local and constrained.* The only long term inelastic deformation remaining is the generalized component creep deformation under virtual steady load conditions.

Failure to meet the requirement of I and II does not mean that the component design cannot be validated at all. What it does mean is that unconstrained inelastic deformations are possible during every load transient, which can add up very rapidly to excess deformations. In principle, validation may still be possible by carrying out a full inelastic analysis of the component using a precise *and well proven* constitutive model of the material response which, however, is very seldom available in practice. Experience so far suggests that if the stage I and II evaluation fails to demonstrate constrained conditions, a component is seldom found to be acceptable after a more detailed inelastic evaluation.

Realistically, this option only exists for very special cases, or when it is necessary to retrieve an earlier design error.

Once overall dimensional stability of the component has been assured in stages I and II, it is permissible to proceed to evaluating material damage at local "hotspots" due to the combined effects of creep and fatigue. Although none of the procedures cited here make any explicit reference to environmental effects, it is recognized implicitly that they do exist, and that their interactions with both creep and fatigue are significant in accumulating high temperature damage. The solution adopted by most design codes and guidelines to use material data collected under similar environments as are experienced in service, so that any environmental effects are inherent in the data.

The following sections outline methods of analysis found generally acceptable for performing the various stages of design evaluation.

A4.2. Stress Classification

Full inelastic analysis is a costly process for many reasons, not least of which is that it cannot be carried out very effectively on systems whose geometries have not yet been tentatively decided upon. For this reason, linear elastic methods are accepted for at least the first cut at design evaluation. There are good reasons why linear analysis still applies to a large proportion of high temperature design, despite the fact that materials are known to behave nonlinearly in this range. These reasons will be given later.

Any valid method is generally accepted for the analysis itself, including hand calculations where standard component geometries are concerned, and Finite Element analysis for more complex geometric shapes.

In order to use linear elastic analysis correctly, it must be recognized that all stresses are not equal. An important distinction is drawn between

a) Primary Stresses
b) Secondary Stresses
and c) Peak Stresses

Primary stresses are derived from external mechanical loads and do not relax under local inelastic deformations. One major category of primary stresses is the *primary membrane* stress in pressure boundaries, or the normal stress on a net section, usually referred to as "Pm", following the ASME Code convention. A second major category is *primary bending, Pb,* e.g. beam bending or circumferential bending in a transversely loaded tube.

Secondary stresses, also known as "Q" stresses following ASME convention, are discontinuity stresses at openings in pressure vessels, or thermal stresses, which are self-equilbrating, and generated entirely by the requirement of enforcing compatibility in redundant or kinematically constrained components.

"F" stresses are local, short range stresses, such as local stress concentrations around holes or notches, or local thermal hotspots on exposed surfaces caused by heat source impingement.

P, Q and F stresses do not all have the same significance in terms of component integrity, which is the reason for making the distinction in the first place. As an example, P stresses are the only ones considered in stage I, primary design, whereas both P and Q stresses need to be considered in stage II, and so on. Details of how stress categories are used in the various stages of the design process will be dealt with in later sections of this chapter.

It should be noted early on, that it is not always obvious how to subdivide the total stress into P, Q and F categories. Consequently, the categorization process forms an important part of the design process itself.

Ironically, stress categorization is an easier task to perform with hand computations than with FEA results. In a hand calculation the P stresses are the "strength-of-material" stresses obtained by fundamental considerations of equilibrium, e.g. "pR/t" stresses in tubes under internal pressure, and "My/I" bending stresses in beams. Q stresses are also relatively easily identified in a hand calculation. They are fixity stresses resulting from imposing compatibility and solving for redundant reactions. In conventional structural

analysis, thermal stresses are obtained by precisely the same procedure of imposing compatibility on specified lack-of-fit displacements, as is used in computing redundancies. F stresses are the local SCF's commonly tabulated for standard structural discontinuities and notches in mechanical design handbooks.

The same easy classification of stresses does not emerge from FEA. The reason is that FEA only provides the *total* stress. It does not distinguish between the root causes of stress. To deal with this problem users of the ASME BPV Code have developed a systematic procedure known as "stress linearization". The complex distribution of stresses on any section are replaced by a statically equivalent linear distribution, in effect the "strength-of-materials" stresses for the same section bending, normal and shear forces.

The linear components of stress are designated P or Q stresses, depending on circumstance. For instance, Pm, the membrane stress is invariably primary. The bending component is considered primary, i.e., Pb, in the circumferential direction in a circular tube, but only secondary, or a Q stress, in the axial direction, because axial bending in a circular tube is self limiting.

The F, or peak stress, is the *excess* of the actual computed stress on the section over the linearized stress.

The ASME classification system has been adopted worldwide. In fact both the British and French design guidelines consistently refer to "P", "Q" and "F" stresses as defined here, as a standard convention, despite the fact that the design methodology in each case is distinct, and both differ in many important ways from that presented in the ASME Code Case N47.

A5. Details of the Design Procedure

A5.1. Primary Design

There are two basic methods in use for assuring primary design limits. The second of these comes in two versions, a detailed method and a simplified one. Selection of an appropriate method depends on the quality of the material data available and the capacity to

perform limited inelastic analyses.

i) ASME Method

Calculate primary load carrying stresses, Pm and Pb, by hand calculation or, alternatively, extract primary stress components from the complete linear elastic stress distribution by the linearization technique.

For acceptance, Pm < Smt

Pm + Pb < 1.5 Smt

ii) Reference Stress Method (Detailed approach)

If a limit load can be computed for the component, the reference stress, σR. os

$$\sigma_R(\text{Strain}) = \frac{Pa}{PL}.\sigma_Y$$

where, $\sigma_R(\text{Strain})$ = Reference Stress

σ_Y = Yield Stress of Elastic, perfectly plastic material

Pa = Actual Applied Load

PL = Limit Load with yield stress, σ_Y

The reference stress, $\sigma_R(\text{Strain})$, is an approximation, and is usually a close upper bound to the exact value. This equation should only be used for strain limits. A modified version of equation 1 should be used for the rupture criterion, i.e.

$$\sigma_R(\text{Rupture}) = \frac{Pa}{PL}.\sigma_y.\{1 + 0.13(x - 1)\}$$

$$x = \frac{(P + Q + F)}{\sigma_R(\text{Rupture})},$$ is the Stress Concentration Factor

For acceptance, σ_R (Strain) < St (1% strain in 10^5 hr)

σ_R (Rupture) < St (Rupture in $3x10^5$ hr)

iii) Reference Stress Method (Simplified approach)

For preliminary calculations the SCF, x, is usually about 2.5, making the term, {1 + 0.13 (x - 1)}, a maximum of about 1.2 An all-purpose reference stress estimate for most practical purposes is therefore,

$$\sigma_R = 1.2 \frac{Pa}{PL} . \sigma_Y$$

This reference stress can then be compared with the design allowable, Smt, without the need to separate it into strain and rupture limits, i.e.

$$\sigma_R < Smt$$

Comparison of Methods:

The ASME stress linearization method is preferred in many applications because it is based on a linear elastic stress analysis. It is not, however, a linear elastic method. The elastic stress distribution is merely used as a one permissible statically admissible stress system, in order to establish a lower bound to the component limit load. The advantages of the ASME approach are

a) The stress analysis can be standardized using existing commercial FEA programs and performed by design engineers without special expertise in nonlinear material behavior.

b) There is virtually no limit on the complexity of the component geometry or the number and complexity of load transients considered in design. For instance, only a few unit load cases need be analyzed fully. All design load cases, including transients, can then be constructed by linear combinations using a much more economical post processing program.

Its disadvantages are

a) It can be difficult at times to differentiate between P and Q stresses. Since

P stresses are subject to the most stringent limitations, this can be excessively conservative if it is not possible to extract the Q components reliably.

b) The method is always inherently conservative because it only recognizes relaxation of stresses on individual sections and does not take into account the ability of a component, such as a plate in bending, to redistribute load <u>between different sections</u>.

The last disadvantage is not as severe as it seems at first sight. In practice the limit load of a component is unattainable because excess deformations intervene long before collapse. The ASME method is a pragmatic way of eliminating local peaks and eliminating internal, self equilibrating stresses while maintaining control over deflections.

It must be recorded that, if it is not possible to satisfy primary limits using the ASME linearization method, the Code still allows direct use of limit load concepts as an alternative. There is no high temperature equivalent to this alternative in the high temperature Code Case N47. However, the Reference Stress technique is an exact analogy to the use of limit load methods for time dependent deformation.

The Reference Stress technique for primary design is the preferred method in the British R5 guideline. Its advantages are,

a) It is more accurate, and not so needlessly conservative, as the ASME linearization procedure because it takes section-to-section redistribution into account. This can be a significant saving in plate or shell structures.

b) The primary stress, <u>is</u> the reference stress. No judgements are necessary to extract P stresses from Q and F stresses.

Its main disadvantages are

a) It is necessary to perform a limit load analysis for each distinct load combination.

b) It does not handle local strain limits very easily.

Both of these disadvantages can be overcome. Limit loads have been computed for

many standard component geometries. Techniques also exist whereby an approximate limit load can be found quite easily for a component of arbitrary complexity using only linear elastic FEA routines, by systematically modifying the elastic stiffness of individual elements. The second disadvantage is automatically eliminated by using the technique of successive elastic stiffnesses just mentioned. This technique is based on successive approximations, approaching the exact solution from below as safe lower bounds. The strains calculated in each approximation provide the exact strain concentration corresponding to the current lower bound estimate.

Conclusions:

Method of primary design evaluation is a matter of choice. The ASME approach is more easily systemized and is probably a good method to use as a first filter. If everything passes this criterion there is no need to get more sophisticated. The reference stress approach works well when the number of load cases is relatively small, when standard limit solutions exist to begin with, and where difficulties are experienced in separating out P, Q and F stresses using the ASME approach. This last problem is particularly difficult to deal with in highly complex three-dimensional components, where simple sections cannot be taken, and there is no prior experience on standard geometries to help in separating stresses into different categories.

A5.2. Shakedown Analysis

Shakedown analysis is protection against incremental deformation, or ratchetting. It is normally considered under two headings, low temperature shakedown and high temperature shakedown.

Low temperature shakedown is shakedown in the original sense of the word, i.e. a steady state in which a constant residual stress system is set up so that all successive deformations are confined to the linear elastic range. In practice this ideal is virtually impossible to attain, because there are always local pockets of inelastic deformation at points of strain concentration. The pragmatic solution to this problem, taken by all existing

design codes and guidelines is to consider only P and Q stresses in evaluating low temperature shakedown.

High temperature "shakedown" is more accurately the avoidance of "creep ratchetting", in other words, accelerated creep deformation due to the interaction between plastic deformation during load transients, and creep deformations in the dwell periods between the transients. A complex component undergoes a period of stress redistribution on first loading which decreases to a minimum deformation rate as the stresses tend toward a steady state. If the progress toward a steady state is upset by plastic deformations caused by a load transient, the steady state needs to be reestablished during the next dwell period before the minimum creep rate is attained once more.

There is no general solution to this problem as yet. Existing codes contain four distinct methods of dealing with shakedown. Each has its own area of application.

i) The "R5" Method

This method applies equally to both low and high temperature shakedown. It ensures shakedown at low temperatures and provides an upper bound on creep deformations under cyclic loading conditions at high temperature.

Let the time dependent elastic stress distribution in the component be Se(t), where Se(t) only includes the P and Q components. Assume any time independent residual stress system, $\bar{\rho}$, to exist in the component. Ways of finding suitable residual stress systems will be deferred for the moment. The total stress at any time, S(t), is therefore

$$S(t) = Se(t) + \bar{\rho}$$

Time independent shakedown will occur if S(t) nowhere exceeds the yield criterion, $\phi(\sigma)$, where $\phi(\sigma_y) = 0$, i.e.

If $\phi(S(t)) \leq 0$ at all times, then shakedown.

When creep deformation is not negligible, creep ratchetting, i.e. interaction between plastic and creep deformations will be avoided if

$$\phi\left\{\frac{(n+1)}{n}S(t)\right\} \leq 0$$

where n = Creep Index in Norton's Law.

An upper bound to the maximum creep strain is obtained by finding the maximum value of S(t) and looking up the corresponding creep strain from isochronous stress/strain data.

This method is based directly on the shakedown concept as formulated by Melan, and extended to high temperature by Goodall et al [9]. It always ensures a conservative solution.

The solution can be optimized by careful selection of the residual stress field, $\bar{\rho}$. In simple examples, $\bar{\rho}$ can be found by inspection. A good value of $\bar{\rho}$ is more difficult to find in the general case.

R5 and UKAEA guideline suggest a systematic procedure for find $\bar{\rho}$ using thermal stress distributions which, by definition, are self-equilibrating. In its simplest form this procedure advocates solving a thermal stress problem, in which pseudo-temperatures are chosen throughout the component to be inversely proportional to the maximum stresses due to combined mechanical and thermal load. The resulting residual stress state, ρT, can now be scaled up or down to maximize the shakedown load, or minimize the maximum creep rate. This last step is a simple postprocessing step, and is therefore very quick and cheap to do.

ii) The RCC-MP Method

This method is based on extensive testing of real components to find shakedown limits experimentally. Unlike the R5 method which, in theory at least, can deal with an arbitrary geometry, the RCC-MP method is restricted to tubelike components of the type

commonly found in fast reactor construction.

iii) The ASME Low Temperature Shakedown Procedure

This procedure applies to any geometry under an arbitrary thermo-mechanical load history, and takes temperature variation of material properties into account as a matter of course.

The Pm+Pb+Q stress intensity range is calculated at all critical points. Shakedown is assured if

$$\text{Max}\ \{(\text{Pm+Pb+Q})\ \text{range}\} \leq 3\ \text{Sm}$$

Since Sm is 2/3 of the yield stress, this constraint implies that the maximum stress range at any point, ignoring the local peaks or F stresses, must not exceed the yield range. Hence shakedown is assured.

This procedure can be completely automated. In fact the ANSYS commercial FEA program has a postprocessing routine to deal it. Unfortunately, there is not, as yet, any high temperature equivalent for this very comprehensive method.

iv) The ASME N47 High Temperature Procedure

The N47 approach to high temperature creep ratchetting is restricted to the Bree problem consisting essentially of a section with a constant, mechanically applied tensile load and a cyclic thermally induced bending moment. The original Bree concept has been expanded by O'Donnell and Porowski, and utilized the concept of an "elastic core". The method has been tested extensively, but is restricted to a very narrow class of problem. There have not serious attempts to generalize the approach to other shakedown problems, probably because it is such a special case.

Comparison of Methods

The RCC-MP and N47 methods are recommended for those special situations where they apply. The RCC-MP method in particular is to be recommended because it contains no simplifying assumptions regarding temperature- and cycle-dependent material properties. These are included inherently in the tests which form the basis of the design changes. However, it is not possible to generalize either RCC-MP or N47 approaches to other situations, including the one commonly found in high temperature applications, in which a large number of different thermo-mechanical transients can occur in virtually arbitrary combinations.

The ASME method based on (Pm+Pb+Q) range is by far the most versatile, and is recommended for low temperature shakedown evaluation. Unfortunately there is no high temperature equivalent of this method.

For general use, when it is not possible to take advantage of special circumstances like those assumed in the RCC-MP approach, the only viable option is the R5 method. In principle, any problem can be dealt with by this method. Its limitations are,

a) It is based on mathematical simplifications of material constitutive behavior which include some radical assumptions about the temperature dependence of material properties. This is the problem the RCC-MP method avoids by making direct use of experimental data.

b) The method is inherently conservative, which is not necessarily a disadvantage, but the quality of the solution depends on how good an estimate of the residual stress, $\bar{\rho}$, can be found. It remains to be seen whether the pseudo-temperature stress method suggested by R5 and UKAEA is still workable when the load histories become more complicated.

c) It is uncertain how conservative the method is for estimating maximum creep strains.

Conclusions

When the special circumstances allow, use the RCC-MP approach.

If only low temperature shakedown is important, use the current ASME BPV Design Code method, especially if many diverse transients have to be considered.

For general high temperature applications, the only viable candidate at this time is the R5 procedure. It may be necessary to simplify the transients in order to make the procedure work.

A.5.3. Creep/Fatigue/Environmental Damage Evaluation

This section covers damage due to fatigue, creep, environment and any interactions between them. It assumes that primary and shakedown evaluations have been performed satisfactorily, and it has been concluded that the component is dimensionally stable. This means that inelastic deformations caused by transients in the load history will be local and constrained. It is assumed that thermo-mechanical transients are short in duration compared with the total design life of the component. This means that the increment of long term creep deformation accumulated during any transient event will be negligible compared with cyclic inelastic deformations occurring due to the transient itself. It is reasonable to assume, therefore, that local transient induced damage mechanisms can be considered as local perturbations on an otherwise *linear elastic system.*

i) Creep Damage

Creep Damage under constant uniaxial stress can be obtained directly from rupture curves or Larson-Miller plots. The problems of interpretation which have to be dealt with by designers damage accumulation under

a) Variable stress/strain history

and b) Multiaxial stress.

Each of these problems has two parts, firstly calculating the time varying stress state at a critical point in a component and, secondly, evaluating the damage caused by the stress history.

Stress/strain analysis

All the codes reviewed allow the possibility of computing local inelastic behavior by making modifications to a linear elastic analysis. This always takes the form of the Neuber notch correction, with minor variations.

The Neuber correction assumes that, in conditions of constrained inelastic deformation on the stress and strain at the critical location vary hyperbolically, i.e.

$$\sigma\varepsilon = \sigma_e\varepsilon_e$$

where,	σ	= maximum inelastic stress
	ε	= maximum inelastic strain
	σe	= maximum linear elastic stress
	εe	= maximum linear elastic strain

When creep and other inelastic deformations are presented in the form of isochronous stress strain curves, it is a simple matter to calculate time-dependent stress-strain behavior from the intersection of the Neuber hyperbola and the isochronous curves.

The Neuber method is only valid as long as the local deformations are constrained. In order to achieve a smooth transition to long term steady creep deformation, the approach used in the ASME Code Case N47 is to make a transition from relaxation along the Neuber curve to the steady load (Pm+Pb) primary stress. This approach is essentially similar in all the alternative methods.

Alternatively, all codes and guidelines allow the option of full inelastic analysis for part or all of the creep deformation problem.

Damage Computation

a) Variable stress/strain history

Despite the development of many more sophisticated methods of evaluating creep damage over the past several decades, the method still used almost universally today is

Robinson's Life Fraction Rule. The only exception is the Ductility Exhaustion Method offered by the R5 guideline as an alternative to the Life Fraction Rule, if the quality of material can justify it.

The Life Fraction Rule asserts that creep damage Di at a given steady state of temperature Ti and stress σi is linearly proportional to the time spent at these conditions, with Di reaching 1 at failure.

$$\text{i.e.}\quad Di(Ti, \sigma i) = \frac{t_i}{t^R{}_i}$$

where, $t^R{}_i$ is the time-to-rupture under (Ti, σi).

Under variable stress and temperature, failure occurs when

$$\sum_{i=1}^{n} Di = 1$$

Ductility Exhaustion is offered as an alternative to the Life Fraction Rule by R5, if appropriate strain-at-fracture data can be found. This information is not generally available for most materials, hence the predominant use of the simpler Life Fraction Rule for most applications.

The principle behind Ductility exhaustion is identical to Life Fraction Damage Summation. Damage, Di, is assumed to be linearly proportional to strain-at-rupture, εRi, under steady conditions of temperature Ti and stress σi, with Di reaching 1 at failure.

$$\text{i.e.}\quad Di(Ti, \sigma i) = \frac{\varepsilon_i}{\varepsilon_{Ri}}$$

where, ε_{Ri} is the strain-to-rupture under (Ti, σi).

Under variable stress and temperature, failure occurs when

$$\sum_{1=1}^{n} Di = 1$$

b) Multiaxial stress

Multiaxial stress state is known to have profound effect on creep damage accumulation, but is not considered in any great detail in any of existing high temperature codes. The reason is that little experimental evidence is available for the majority of materials. Most of the time there is not much more the conjecture to go on.

The two primary candidates for a governing criterion for damage accumulation under multiaxial stress are the maximum principal tensile stress, and the effective stress (either Mises or Trasca). It is known that materials in general are governed by both of these criteria, but depending to different degrees on each. For instance copper is predominantly a "Maximum Principal Stress" material, whereas aluminum alloys tend to be "Effective Stress" materials. Most practical engineering alloys tend toward an effective stress dependence rather than principal stress dependence. This is a matter of natural selection, because "principle stress" materials are overly sensitive to hydrostatic stress and therefore very sensitive to notches.

ASME Code Case N47 reduces all stresses to the so-called "stress intensity" which, in Code terms, means the Tresca stress. By default, therefore, this guideline assumes that the governing stress criterion for damage accumulation under multiaxial stress is an effective stress, similar to the criterion for plastic deformation.

The same default position is taken by other codes and guidelines. Exceptions to this general rule are specific materials, notably Type 316 stainless steel, which has been tested extensively because of its use in fast reactor construction. In this case the simple effective stress model is adopted for *tensile stress* but it is assumed that no damage is accumulated under *compressive stress*.

At present there is no general rule on how to deal with individual materials, or how to screen them quickly to determine tendencies.

Comparison of Methods

The Life Fraction Rule has been used for more than 40 years. It has many detractors, but endures because nothing better has been found which works as well under all circumstances, and with the quality of information designers have to cope with most of the time. This may be the only method available in many instances.

The only serious contender with any prospect of challenging the Life Fraction Rule is Ductility Exhaustion. Intuitively damage appears to be linked more rationally to strain than to time-at-temperature. However, care needs to be taken in defining the "strain" component to be equated with damage. For instance, the early stages of primary creep is not as damaging in the creep rupture sense as later steady state creep. It is tentatively assumed that the main damaging component of creep deformation is the Monkman-Grant creep strain, εmcr, given by

$$\varepsilon_{mcr} = \dot{\varepsilon}_{mcr} t_R$$

As far as multiaxial stress is concerned, the only generally usable hypothesis seems to be the same effective stress model used to describe plastic deformation, whether this be Tresca or Mises.

There is no doubt that a more detailed model, taking mean stress effects into account, would be a great improvement, but the experimental data needed to fit the model is seldom available. In the meantime, the best that can be done is to distinguish between net tension and net compression, and assume that creep damage only occurs under tensile conditions. It should be noted though that there are mechanisms of creep damage which do not involve cavitation damage. These materials damage at the same rate in tension and compression.

ii) Fatigue Damage

Fatigue damage in the absence of any time dependent effects is generally adequately

described by the Manson/Coffin/Basquin Equation,

$$\frac{\Delta\varepsilon_t}{2} = \sigma_f^1(2N_f)^b + \varepsilon_f^1(2N_f)^c$$

Damage per cycle is computed for a given cyclic state at a given temperature ($\sigma_{i,}$ Ti) as

$$dDi\,/_{dN} = 1\,/\,N_{fi}$$

iii) Creep/Fatigue/Environmental Interaction

There are at least two options available for dealing with creep/fatigue interactions. One is a very simple method. The second is a much more sophisticated approach, using one of several complex constitutive models of behavior.

a) Simple Linear Damage Summation

Linear Damage Summation calculates the creep damage, Dc, using either Robinson's Life Fraction Rule, or Ductility Exhaustion, from the stress/time history of the variable load cycle. It then calculates fatigue damage, Df, for the stress/strain cycles in the load history. Failure is assumed to occur when the total damage, D, reaches a critical value

$$D = Dc + Df$$

The simplest version of this method assumes at failure that D = 1. This has since been modified for certain specific materials to take account of a more severe interaction than a purely linear one. For instance, ASME Code Case N47 and other design guidelines assume a bilinear interaction, Where $D \rightarrow 1$ if Dc>>Df or Df>>Dc, and D = 0.6 if Dc=Df.

b) More Detailed Options

There is a wide range of detailed creep/fatigue/environment interaction theories in existence to choose from. The better known ones are

Frequency Modified Fatigue

Strain Range Partitioning

Mechanistic Damage Method

Continuum Damage Based Methods

Fracture Mechanics Based

Comparison of Methods

Each of the methods just mentioned has its advocates - and its detractors. The result of many comparative studies is inconclusive. Each method appears to be better than the others in the hands of its originator. In independent studies, however, on anything but carefully selected samples of pedigree material, none of the sophisticated methods perform consistently better than the simple Linear Damage Summation Model, or its bilinear modification for some materials.

Continued use of the Linear Damage Summation Model is not therefore an act of ignorance. Often the code formulators are developers of creep/fatigue models themselves and are well aware of progress - or possibly lack of it - in this area. Linear Damage Summation has survived because nothing has been found that is consistently better, or is able to work at all, given the quality of information that is normally available to designers.

Conclusions

Use Bilinear Damage Summation as tending toward conservatism for most materials, at least for preliminary design.

If problems are encountered using simple damage summation the use of one or other of the special models may be justified, recognizing that this will probably mean a special experimental program.

The one case where use of special models may be worthwhile is where extreme circumstances exist, which are also fairly precisely known, and can be simulated reasonably accurately in a laboratory. One such problem area is so-called *thermo-*

mechanical fatigue. This is cyclic damage under large variations of temperature, so that the material properties vary radically around the stress/strain cycle. For instance, a carbon steel may actually change phase from ferritic to austenitic during the cycle. There may be some justification in cases such as this, where several damage mechanisms are acting simultaneously and interactively, to compute the damage incrementally around the hysteresis loop. This procedure has been proven practical, if computationally very expensive, by Sehitoglu and his students [7].

A5.4. Defect Tolerance

Only the British R5 guideline presents any method for dealing with defects, whether pre-existing or defect which form in service.

Evaluation of significance of defects at low temperature has been documented very well in the CEGB R6 Rev3 document and will not be discussed further here, except to say that this method is accepted worldwide, and is used routinely by organizations such as Westinghouse, Babcock and Wilcox, EPRI and the USNRC.

The only new problem that needs to be dealt with at high temperature is the phenomenon of creep cracking under steady load. R5 presents a comprehensive procedure for calculating the parameters governing creep crack initiation and growth. This procedure is lengthy and no attempt will be made to summarize it here, except to point out its main innovations.

The only new concept required in going from low temperature facture to creep fracture is the C* parameter. C* is the creep equivalent of J for time independent plasticity.

One innovation contained in R5 is a general method for computing C* for cracks in complex geometries and stress/strain fields.

$$C^* = \sigma_{ref}\dot{\varepsilon}_{ref}R$$

where $$R = \frac{K^2}{\sigma_{ref}^2} = \frac{\alpha^2\sigma^2\pi a}{\sigma_{ref}^2}$$

α = Geometry/stress dependent constant in LEFM computation of K

For a constant strain field this method reduces to me formula

$$C^* = \sigma_{ref}\dot{\varepsilon}_c \pi a$$

where σ_{ref} = the Reference Stress

$\dot{\varepsilon}_c$ = creep strain rate under stress of σ_{ref}

The second innovation is an approximate expression developed by Webster for the relationship between C* and the crack growth rate, da/dt.

$$\frac{da}{dt} \approx \frac{0.3(C^*)^{\frac{n}{(n+1)}}}{\varepsilon_f^*}$$

where ε_f^* = creep ductility corrected for triaxiality

Uniaxial ductility, ε_o, is related to multiaxial ductility by the Rice/Tracey formula,

$$\frac{\varepsilon_f}{\varepsilon_o} = 1.65 \exp\left(-\frac{3\sigma_H}{2\sigma_{eH}}\right)$$

According to this formula, the ratio of ductility at a sharp crack tip in plane strain to plane stress is about 1/50. Plane strain cracks have been shown to propagate at approximately 50 times the rate seen in under plane stress conditions.

Appendix B - Some Specific Aspects of the Freedom Space Station Solar Collector Design

Introduction

These comments follow approximately on the format of the generic design guideline attached. Numbers in parentheses refer to the section numbers on the accompanying design guideline given in Appendix A. Wherever possible, each comment is accompanied by a "General Recommendation," "Further Analysis," and "Material Testing Required." Other than this loose structure, no order has been placed on the comments.

(A1.1. Tensile Properties of HAYNES 188)

Problem:

Primary stresses in the structure are low (< lksi). This appears at first sign not to be a problem. However, the allowable St for Haynes 188 is also likely to be low. There is no test data extending out to 260,000 hours so extrapolation techniques are necessary to estimate St at the design life by indirect means.

Simple methods, such as the Larson-Miller Parameter, have been used to find theoretical St values. These have yielded best estimate St values between 5.5 ksi and 10 ksi at 260,000 hours and 1400^{o}F. IT is reasonably well know that simple extrapolation methods can be optimistic because they do not capture the early onset of damaging effects which only show up clearly after long times on test.

A combination of lower bound (e.g. 90% LCL on data), and a more sophisticated extrapolation technique could easily bring the allowable St values down to the primary stress levels calculated in the system.

Recommendation:

At least 2, and possibly more, more sophisticated extrapolation methods, such as Manson's Minimum Commitment Method, and the Wilshire-Davies θ-Technique should be used to construct more reliable long-term rupture and strain design curves from existing creep data.

Further Material Testing:

The quality and amount of existing basic creep data on Haynes is superior to the data available on most materials intended for long-term, high temperature service. It also extends up to within a factor of 10 of design life. It should be possible to construct reliable design data without further testing in this area.

(A1.2. Creep Data)

Problem:

Creep data in the form of strain/time curves has limitations in design, especially under variable stress and relaxation conditions. This is a problem where the majority of creep deformation and damage is likely to be accumulated under almost pure relaxation conditions.

Recommendations:

Available creep deformation data should be replotted in the form of isochronous stress/strain curves. In this case both forward creep and relaxation data at fixed strain have been collected on Haynes 188. Both sets of data should be used, making certain to include any short-term inelastic deformations occurring on first load application.

It should not be surprising to find that forward-creep and relaxation isochronous curves are different. If Haynes 188 follows the normal trend, relaxation curves will be lower than forward creep curves. However, the two sets of data can be used, with a little interpretation, to calculate time dependent deformations very satisfactorily, without recourse to complex constitutive modelling.

Further Material Testing:

There is more creep and relaxation data on Haynes than can be found for most high temperature materials. With the exception of "fill-ins" revealed when the available data are plotted as isochronous curves, there is no desperate need for more testing in this area.

(A1.3. Fatigue Data)

(A1.3. Fatigue Curves)

(A1.4. Creep/Fatigue Data)

Problem:

Nominal strain ranges. at least in the canisters, are small compared with the short-term elastic range of this material. Furthermore, the thermal range in the normal 90 minute cycle is not large enough to evoke questions of thermo-mechanical fatigue in the sense of radically varying material properties around the stress/strain cycle. It seems, therefore, that fatigue *per se* is not a problem in the absence of defects.

However, this application qualifies as one of "Ultra High Temperature Design." This means that the material long-term creep strength, ST(~ 5 ksi, est.) is about an order-of-magnitude lower than its short-term yield strength (~ 40 ksi, even at peak temperature). There must be some concern over a possible interaction between fatigue cycles and ongoing creep.

Unfortunately, all the available fatigue, creep/fatigue and thermal fatigue data on Haynes 188 is for much more severe conditions of strain and temperature range than is experienced in those parts of the solar collector examined so far. Attempts have been made to extrapolate the existing fatigue data to higher lives and less severe conditions without success. Even if numbers could be obtained by some extrapolation technique they could not be trusted.

Recommendation:

The precise form of the expected stress/strain/time cycles for critical components of the solar collector need to be studied more critically and a case-specific procedure developed for assessing cyclic damage. A tentative proposal follows.

a) Canisters: Since the computed stress range in these units is less than 1/2 the short term yield range of the material, it appears that there will be no resetting of stresses by plastic deformation at the cold end of the cycle. Therefore, the total deformation process

can be separated out into

i) a time independent inelastic stress/strain loop

ii) a single creep relaxation under steady insolation conditions, lasting the entire design life.

The only opportunity for creep/fatigue interaction is in the first few tens or hundreds of cycles, when relaxation is occurring at a perceptible rate during the hot end of the cycle. It is expected that this should be negligible. Creep/fatigue interaction during this part of the service life is ideally suited to computation using Strain Range Partitioning (this is a special circumstance which may not happen very often). Once relaxation is complete, there should be no further interaction unless residual stresses are reset during cold shutdowns.

Further Material Testing:

The expected cycles in service are so far removed from existing experimental data that it is important to do a few, carefully chosen cyclic tests that are more typical of real service conditions.

e.g. Given the programmable capability of modern test facilities, it should not be difficult to simulate the initial, relaxation phase of service conditions, with a programmed reducing mean stress and a relatively low stress cycle superimposed. It is obviously impractical to simulate service conditions exactly, but the damage potential of the initial relaxation phase could be tested by changing to rapid cycle fatigue once creep relaxation has ended.

NOTE: There are other parts of the solar collector, such as manifold/tube intersections, where fatigue is likely to be an even bigger problem than in the canisters. Even there, however, strain ranges are likely to be radically smaller than those used so far in creep/fatigue and thermal fatigue testing, so there is a definite lack of data in this area.

(A2. Definition of Loadings)

Problem:

The design evaluation of the solar collector has focussed on the canisters to the apparent exclusion of all other components. There are other loadings on the structure as a whole which may be more significant than the thermal cycle experienced by this particular element.

There should be some cause for concern over the inordinate amount of computational effort applied to the canisters, in view of the other important system loading which seem to have been neglected as a result. In passing it should be mentioned that the FEA results on the canister can be reproduced relatively easily using simple hand calculations. On the other hand, the stress analysis of the solar collector as a whole is one applications where a sound FE model really can be justified.

This note does not presume to enumerate all the other loadings to be considered but highlights just a few in order to make the point that a much more comprehensive survey needs to be done to ensure that all the load cases to be experienced in service have been identified.

i. Primary pressure loading of the heat exchanger loop. Primary stresses in this loop are low, but may still be critical given that more stringent limits have to place on primary stresses than on secondary (i.e. thermal and fixity) stresses. It is not clear, until a more detailed extrapolation has been done, that this circuit will, in fact, support the sustained pressure loading for 260,000 hours of operation.

ii) Although assurances have been made that solar heating of canisters on one side only produces negligible thermal gradients, the small gradients which are produced nevertheless cause thermal bending of the main heat exchanger tubes (temperature differences read, with difficulty, from copies of FE output). This in turn causes rotational bending of the tube/manifold connections and, on the basis of a simple hand calculation using formulas from the Kellogg Piping Handbook, produce primary

membrane stresses alone in the connections of 11.6 ksi! This is nearly four times higher than the approximately 4 ksi of secondary bending stress computed in the canisters. Local Q and F stresses in these tube/manifold connections, required for full fatigue analysis, could be two to three times larger than 11 ksi.

iii) Accident Cases - It is not unlikely that one canister will be lost out of approximately 6000 in the course of 30 years of operation. This may not be as catastrophic as the rupture of the underlying tube, due to the change in local heat transfer properties. A simple hand calculation suggests longitudinal bending stress set up in the tubing under an evacuated canister of 20 ksi.

Other cases identified but not analyzed in detail are

i) Tube/manifold connection stresses due to differential longitudinal thermal expansion of tubes caused by off center insolation under normal operating conditions.

ii) As above, but for faulted insolation, e.g. loss of focussing control.

iii) Stresses in bolted connections between main circuit and support casing.

iv) Effects of incomplete brazing - canister closure

-

canister to tube

v) Severity of cyclic inelastic deformation during transients other than the 90 minute orbit, e.g. stoppages for repair and maintenance (if any), any emergency trips initiated by the rest of the system. If any do exist will they reset the residual stresses in the canisters to that the long-term startup relaxation transient is repeated.

Recommendation:

Before any further studies are made of component details, a reasonably accurate, linear elastic FE analysis should be carried out on the solar collector as an integral unit.

This is not an unrealistic task. FE analyses of similar complexity are developed routinely in design of conventional components such as pressure vessels for petrochemical applications and mechanical engineering machinery. The cost of an analysis of this type will probably be less than the detailed FE work done so far on the canister alone.

Further Analysis:

Standard linear elastic 3-D model using standard commercial software (e.g. ANSYS). Use simplified geometric model, e.g. beam elements for tubes and coarse shell elements for manifold. <u>Identify and model all transients</u>, and calculate system loads accurately.

Once there is a clear idea of where stress "hotspots" exist in the system these can then be analyzed in more detail under local loading.

Further Material Testing: None needed.

(A1.3. Design Procedure)

A1.4. Basic High Temperature Design Methodology)

(A1.4.2. Stress Classification)

(A1.5.1. Primary Design)

Problem:

The design of the solar collector needs to be systematic, following procedures of the kind described in Appendix A.

The desire to avoid inelastic analysis is understandable and one which is shared by most design engineers dealing with high temperature problems. It is for precisely this reason that it is important to follow a well established routine which has been validated after a considerably amount of experience. Following the established methodology in broad terms ensures, among other things, that those problems which can be evaluated using linear

elastic analysis are evaluated elastically. It shows clearly where the elastic route breaks down, where inelastic methods need to be employed, if at all, and in precisely what capacity.

A factor which is particularly important in designing a complex component of this nature is have a clear idea at the outset of the concept of stress classification. Many of the concerns raised in the thermal analysis of the canister, for instance, fall away once the secondary nature of the stresses is realized.

Recommendation:

Repeat the analysis of the solar collector, from the top down, using the methodology supplied.

Further Material Testing: None to deal with this point.

Further Analysis:

Standard "ASME Section III" type stress analysis of the complete solar collector system.

(A1.5.2. Shakedown Analysis)

Problem:

The question of whether the canister creep strains are going to accumulate indefinitely is a matter of establishing the criterion of shakedown before proceeding to evaluating material damage.

Clearly, in this case, the only stresses in the canister are Q and F stresses. These are self-limited, by definition, and creep deformation is therefore predicted to reach a maximum limit.

This problem is one which is ideally set up for using the R5/UKAEA method of assuring shakedown. It can be seen that the appropriate residual stress distribution is the maximum thermal stress. If this is superimposed on the time dependent elastic stress

history, the maximum stress at the hot end of the cycle is virtually zero. The upper bound on steady, long-term creep is therefore also zero!

The only proviso to add to this prediction is that it is an extreme application of the theoretical model which forms the basis of the bounding technique. If the thermal transients were sufficient to reset the maximum creep stresses at the beginning of each cycle, the method would probably be more reliable in predicting the resulting drift. In this case, however, there is not resetting. Each hot cycle takes up where the previous one left off. Unless other transients exist which reset the canister stresses, the creep of the canisters is essentially one, continuous relaxation under effectively steady load.

The canister load is analogous to cooling of a metal tray when removed from an oven. There is no primary load to force deformations in any specific direction, but it is possible that some distortion can be developed during relaxation creep due to the "floppy" nature of the structure. This is one inelastic computation that can be done relatively easily, and appears to be worthwhile.

Further Analysis:

One no load condition, steady state relaxation calculation for a canister, with initial residual stress state equal to the maximum thermal stress distribution experienced during the 90 minute orbital transient, is all that is necessary to completely define the time dependent distortions of the canister.

Assuming that no other, more severe, upset or accident transients can be found which reset the residual stresses, the deformations extracted from this single inelastic analysis will be the asymtotic deformations at the end of the canister life.

If other reset transients are found, the canister distortion will be approximately those calculated for a single relaxation multiplied by the number of reset transients.

Recommendation:

The creep relaxation just described does not involve an excessive amount of analysis. It is strongly recommended that it be carried out.

Since the contractors have already developed a model including creep deformations, this analysis should not take long to complete.

(A1.5.4. Defect Tolerance)

Problem:

Creep and fatigue in the nominal (i.e. defect free) structure are believed not to be critical, although there are still some high stress locations which need to be checked out.

On the other hand, there is a much higher chance that failure of wither the canisters, or the primary circuit, might be breached by failure initiated from a pre-existing defect.

Unfortunately there is no crack growth data available for Haynes 188, at least not short crack data of the type that is likely to be relevant to this kind of thin walled structure.

Assuming that Haynes 188 is not a pathological material which differs radically from the other alloys of similar strength, it is possible to make some scoping type estimates of the probable risks of failure due to pre-existing defects. The following calculations are for demonstration purposes only, to determine whether there is a problem or not. They do not purport to predict crack growth accurately, nor do they cover all eventualities.

Assuming a maximum stress in the canister of ~ 3 ksi, and using data from R5 for 316 SS at elevated temperature, in the absence of specific Haynes 188 data, a half-thickness surface crack is computed to grow

$$\Delta a \text{ (creep)} = 3.3 \text{ mils in } 260{,}000 \text{ hours}$$

$$\Delta a \text{ (fatigue)} = 16 \text{ mils in } 260{,}000 \text{ cycles}$$

The analysis these numbers are based on is insufficient to determine definitely whether initial defects will be a problem. They do indicate that they could be a problem, since there is no guarantee as yet that the canister is the most critical element.

Further Materials Testing:

If more accurate crack growth data can be found, it will then be worthwhile doing a more detailed crack growth analysis. At the present time or, if no better material data can be found in the future, there is no point in doing further analysis. The essential point has already been established, that growth of preexisting defects appears to be a definite risk.

Recommendation:

An experimental program to determine growth of small defects in Haynes 188 is strongly recommended if this project is to go ahead.

It may be argued that the problem can be eliminated by preservice testing and repair of all preexisting defects. In practice NDE has a finite probability of failure to detect defects, and a particularly poor reliability in finding close gap, or cracklike defects. It is believed that this structure will almost certainly have at least one undetected defect present if and when it goes into service. Based on the experience of the nuclear and aerospace industries to date, it is likely that the number of defects will be larger, rather than smaller, than any estimates made before launching. This has always been the case in the past and there is no reason to believe it will change on this project.

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